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Experimental and Theoretical Studies of Capillary-Pumped Loop Heat Pipes

1. Introduction

The objectives of this research are two-fold; one is to carry out an extensive experimental investigation of capillary-pumped loop (CPL) heat pipes with single and multiple evaporators. The second objective focuses on analytical studies, the ultimate goal of which is to develop and validate a computer model that can be used to predict the performance of CPL heat pipes over a wide range of conditions. A discussion of the tasks undertaken during the first semi-annual period and an outline of the future plans are presented below.

2. Project Status

2.1 Literature Search and Review

The ongoing efforts in the literature review are intended to broaden the PI's research background on the heat pipe design and technology. A baggage of published material has been collected, including two recent books on the subject (Dunn and Reay, 1994; Silverstein, 1992).

This review of the published material reveals several things. First, there has been almost no experimental investigation of the effect of the vapor groove geometry on the CPL heat pipe performance. The groove geometry is an important feature of any CPL heat pipe design for variable heat load applications. For instance, normal operation of the heat pipe may be realized at high heat loads with a given vapor groove geometry; however, the wick could dry out in the evaporator region at low heat loads because of insufficient driving force for vapor transport to the condenser. Also, the literature data were not presented in forms that can be readily applied to design purposes. There is the need for a critical review of the literature including re-analysis of some of the published data.

2.2 Numerical Study

A numerical study was carried out to determine the effects of geometric design parameters on the CPL heat pipe pressure drop and heat transfer characteristics. The results of this analysis are intended to aid in the selection and design of the configurations to be tested experimentally.

The model equation for the pressure drop calculations is the following dynamic equilibrium relation:

$$\Sigma h_D \geq (\Sigma h_f) w^2 \tag{1}$$

where Σh_D is the total driving head and $(\Sigma h_f)w^2$ is the total system pressure loss; Σh_f is the frictional pressure coefficient and w is the mass flow rate. Several versions of a computer program were written to effect the calculations. The pressure drop results are presented and briefly discussed here. Work on the heat transfer aspect of the analysis is in progress.

Prior to the discussion of the results, there are several general comments. First, all calculations were carried out for a fixed ammonia saturation temperature and pressure corresponding to 25°C and 160 psia, respectively. Also, the vapor and liquid transport lines, each 3.1 m in length, the form loss coefficients due to fittings, the condenser length (1.8 m), wick permeability and pore radius were the same for all calculations. It is worthy of note that, in its present form, the program can handle changes in these variables.

Vapor groove geometry

The shape and governing dimensions of the vapor passages can have profound effect on the CPL heat pipe performance, especially at low heat loads that result in small values of vapor groove pressure drop. For the CPL tested at the Goddard Space Flight Center (GSFC), the vapor passages were trapezoidal in shape. Fluid mechanics and thermal issues favor an exploratory study with triangular passages, hence such an analysis was attempted. Comparison of the pressure drop characteristics for trapezoidal and triangular-shaped grooves are given in Figures 1 and 2; the bases of the comparison are the same total flow area and the same number, N (=45), of vapor flow passages.

For the sake of clarity, the calculated values are indicated using symbols in these figures and some of those presented subsequently. The reason is that fitted curves through the calculated values did not give accurate trends over the entire heat load range. Also, the total pressure drop is the sum of the contributions due to the vapor groove, the vapor and liquid transport lines, condenser, the liquid core within the evaporator and that due to fluid acceleration.

To effect the pressure drop calculations with triangular passages, the pressure drop results of Carlson and Irvine (1961), Hanks and Brooks (1970), Hanks and Cope (1970) and Cope and Hanks (1972) were used. The calculations were carried for seven values of apex angles, α (= 4.01, 7.96, 12.0, 22.3, 38.8, 45 and 60 degrees) for which verifiable pressure drop data are available. The GSFC trapezoidal geometry was approximated as a rectangular channel having an aspect ratio of unity; the literature pressure drop results for this geometry was used.

For a range of evaporator heat loads, Q, the trapezoidal and the triangular geometries give about the same vapor groove or total pressure drop; the latter gives somewhat higher values at large heat loads. The most interesting result is that the change in slope occurs at a lower heat load with the triangular geometry. This change occurs at a vapor groove Reynolds number of about 1600 ($Q \approx 1200$); the corresponding Reynolds number with the trapezoidal cross-section is about 2300 ($Q \approx 1600$). This is a reflection of early transition from laminar to turbulent flow with triangular channels than with rectangular or circular geometry (Carlson and Irvine, 1961; Eckert and Irvine, 1956).

Because laminar flow heat transfer potential differs vastly from its turbulent flow counterpart, the implication is that triangular grooves may be superior to the rectangular geometry. This observation is reinforced by the well established fact that laminar and turbulent flow co-exist in narrow triangular passages even at low Reynolds numbers (Eckert and Irvine, 1956; Bandopadhayay and Hinwood, 1973). So, the existence of the two flow types for vapor groove Reynolds numbers less than 1600 for $Q < 1200$ provides an attractive feature that cannot be realized with trapezoidal or rectangular grooves.

Circular versus rectangular evaporator

For some space applications, it appears that benefits may be realized by replacing tubular with rectangular (or flat plate) evaporators. Some of the advantages include wide surface area for liquid distribution and large contact area for heat transfer. Also, the rectangular evaporator can serve as a baseboard for instrumentation.

Numerical simulations of the CPL heat pipe with cylindrical and rectangular evaporators were carried out for a range of heat loads. The bases for the comparison are the same length and cross-sectional area; these amount to maintaining about the same wick flow area-to-wick volume ratio. Also, the effect of reducing the wick thickness or varying the wick flow area/wick volume ratio was considered. In these calculations, the triangular cross-section of the vapor grooves was of the equilateral geometry.

The calculated pressure drop characteristics are given in Figures 3 through 5. For the circular geometry, the numbers in parentheses are, beginning from left to right, the wick outside diameter, inside diameter and thickness. Similarly, the numbers correspond respectively to the wick outside width, inside width and thickness for the rectangular geometry; these definitions are given in Figure 6 for the 24.7 mm wide rectangular geometry. For the results in these figures, the values of the wick flow area/wick volume are 0.12, 0.14, 0.19 and 0.26 $(\mathrm{mm})^{-1}$ for the circular, 24.7, 37.5 and 50.3 mm wide rectangular geometries, respectively.

For the same wick flow area/wick volume ratio, the wick, vapor groove and total pressure drop are about the same for both cylindrical and rectangular evaporators. This observation should not be viewed as implying that there is no advantage of one configuration over the other because, as noted earlier, there are practical considerations that should be taken into account. The effect of increasing the wick flow area/wick volume ratio or reducing the wick thickness is significant; the maximum operating limit can be increased by as much as 20 percent based, of course, on the conditions given in the figures.

Number of grooves and apex angle

The effect of fixing the total vapor groove flow area and varying the number (N) of grooves is illustrated in Figures 7 and 8, while the influence of the apex angle is shown in Figures 9 and 10. Clearly, the vapor groove or total pressure drop (excluding the wick pressure drop) decreases with decreasing N, resulting in modest improvement in the heat transport capability. So, reducing the number of grooves produces the same effect as that obtained by reducing the wick thickness. Because the vapor groove Reynolds number increases with increasing N, the indication is that early transition may also be realized with fewer flow passages flow passages relative to $N = 45$.

With regard to the influence of the apex angle, the general trend with increasing apex angle from 4.4 to 60 degrees is one of decreasing pressure drop for a given heat load. From these results the following observations can be made. For applications involving low heat loads (up to 1000 W), narrow isosceles triangular grooves (apex angles in the 4 to 40 degree range) are to be preferred. For large heat load applications, the equilateral geometry may be appropriate.

In summary, the highlights of the project status during the period covered by this report include:

- acquisition and review of the relevant literature; a search of the literature failed to reveal any investigation of the effect of the vapor groove geometry on the CPL heat pipe performance.
- a numerical study of the vapor groove geometry for a range of heat loads.
- comparison of cylindrical and rectangular evaporators.

3. Under-Represented (Minority) Graduate Research Assistants

In addition to contributing directly to NASA research and enhancing the PI's research capabilities in areas of interest to NASA, the goal of the program was to provide under-represented minority graduate and undergraduate students who are U.S. citizens with research experience in areas relevant to NASA's mission.

A senior mechanical engineering student, Roger A. Green, was working part-time on the project during the school year, and was appointed a Research Trainee for the summer. Although he obtained his BS degree in May 1994, he was not accepted into the graduate program due to his academic records. His appointment ends on September 26, 1994.

Attempts have been made to recruit minority Graduate Research Assistants. In December 1993, Dean Suzanne Liberty of the Graduate School contacted Mr. Robert Lewis of the National Consortium for Graduate Degrees for Minorities in Engineering and Science, Inc. (GEM) who circulated our advertisement to a pool of students. In March 1994, the PI represented the Graduate School and the School of Engineering at a Career Fair organized by the National Society of Black Engineers' (NSBE) in Pittsburgh and met with many graduating seniors. The PI also contacted colleagues at Historically Black Universities (North Carolina A

& T and Howard) to help identify potential graduate school applicants. Suffice it to say that these efforts have not resulted in the recruitment of a full-time graduate research assistant for the project.

4. Future Plans

The goals for the next five months are summarized below.

- Acquisition of the necessary equipment and supplies, and the construction of the test-loop for the single evaporator/condenser studies. The fabrication of the evaporators with triangular vapor grooves will be made in consultation with Dr. Jentung Ku and Mr. Ted Swanson of the Goddard Space Flight Center.
- To continue with the modeling efforts, specifically, the heat transfer analysis and the development of transient model.
- To work closely with the Graduate School and the Graduate Admissions Committees of the chemical and mechanical engineering departments to recruit full-time research assistants for the project.

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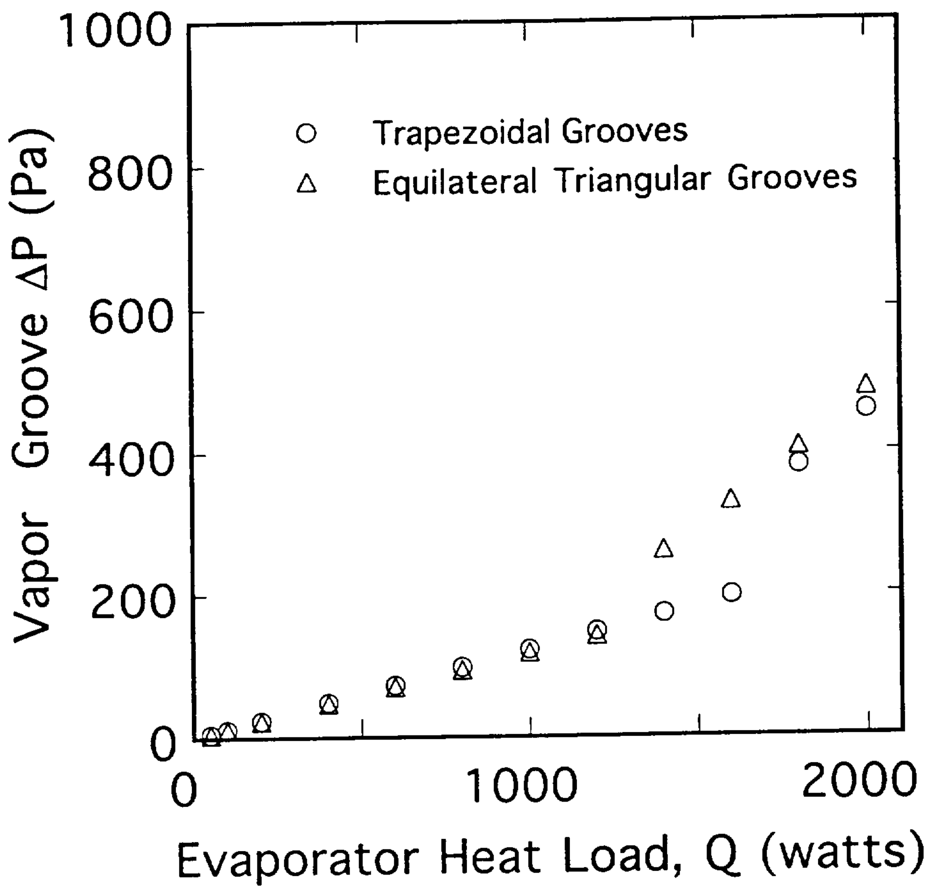

Figure 1. Vapor groove pressure drop for trapezoidal and triangular passages.

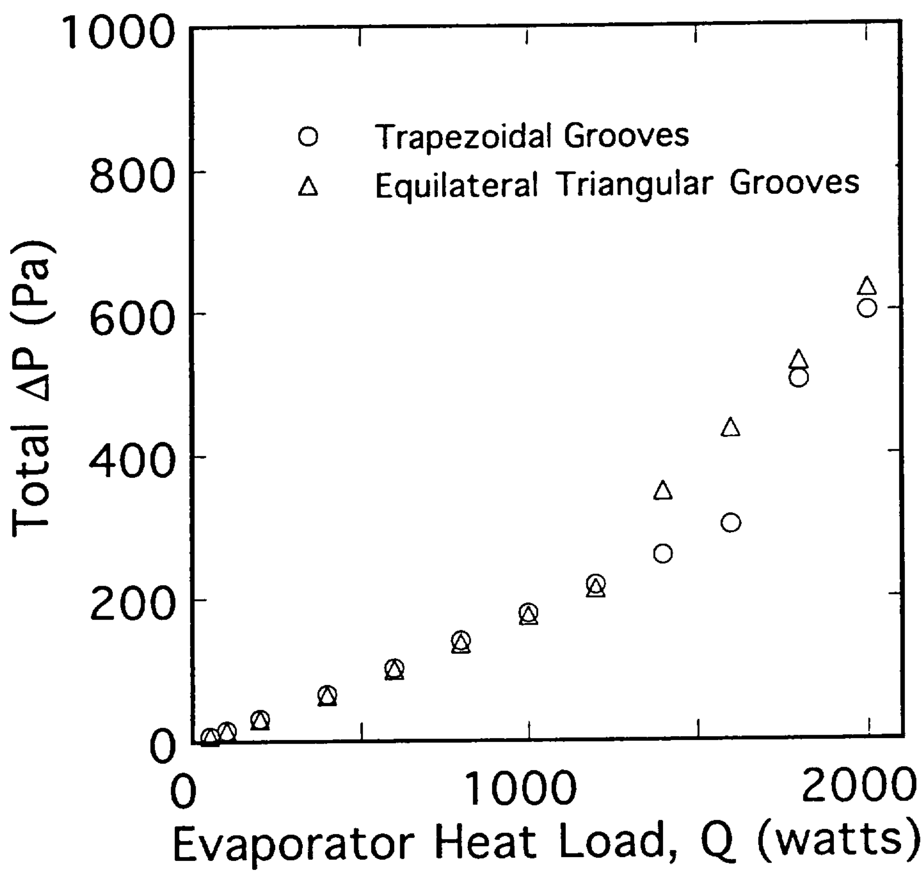

Figure 2. Total pressure drop for trapezoidal and triangular passages

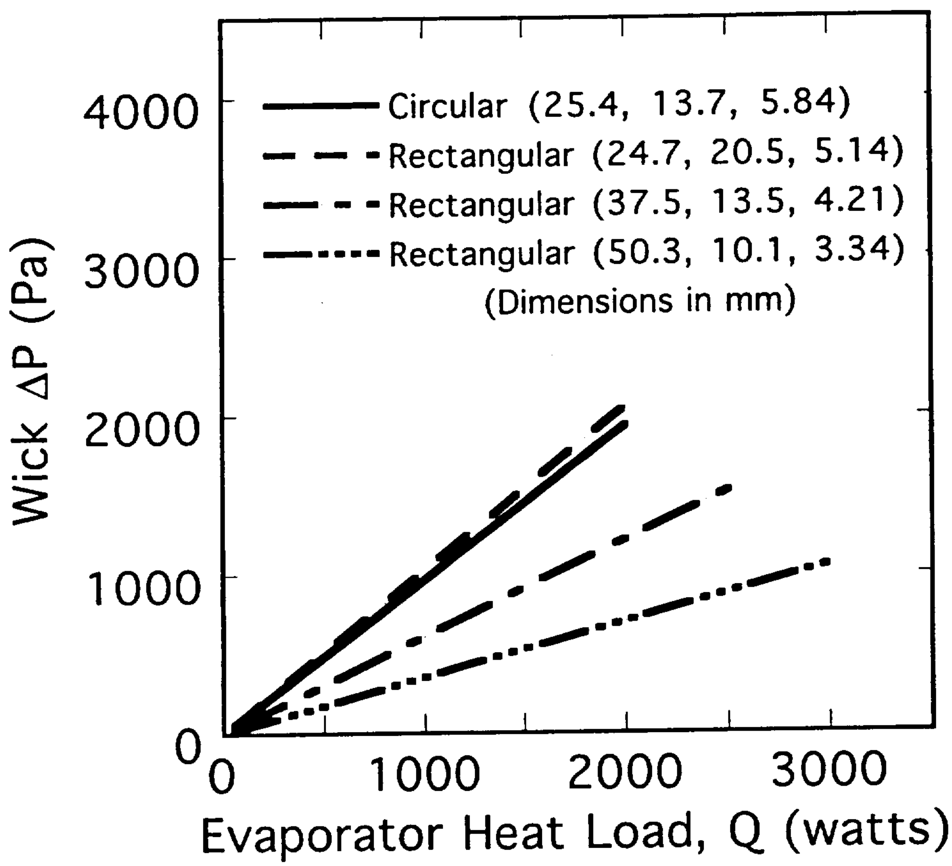

Figure 3. Pressure drop for cylindrical and rectangular heat pipes.

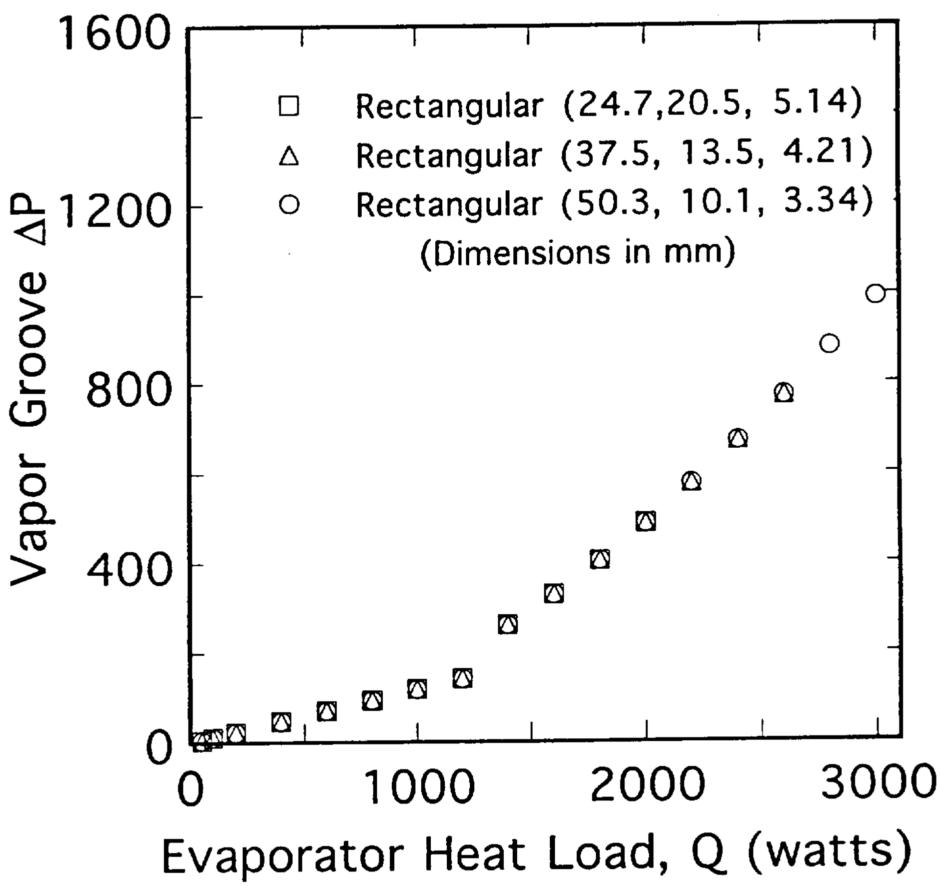

Figure 4. Effect of wick thickness on vapor groove pressure drop.

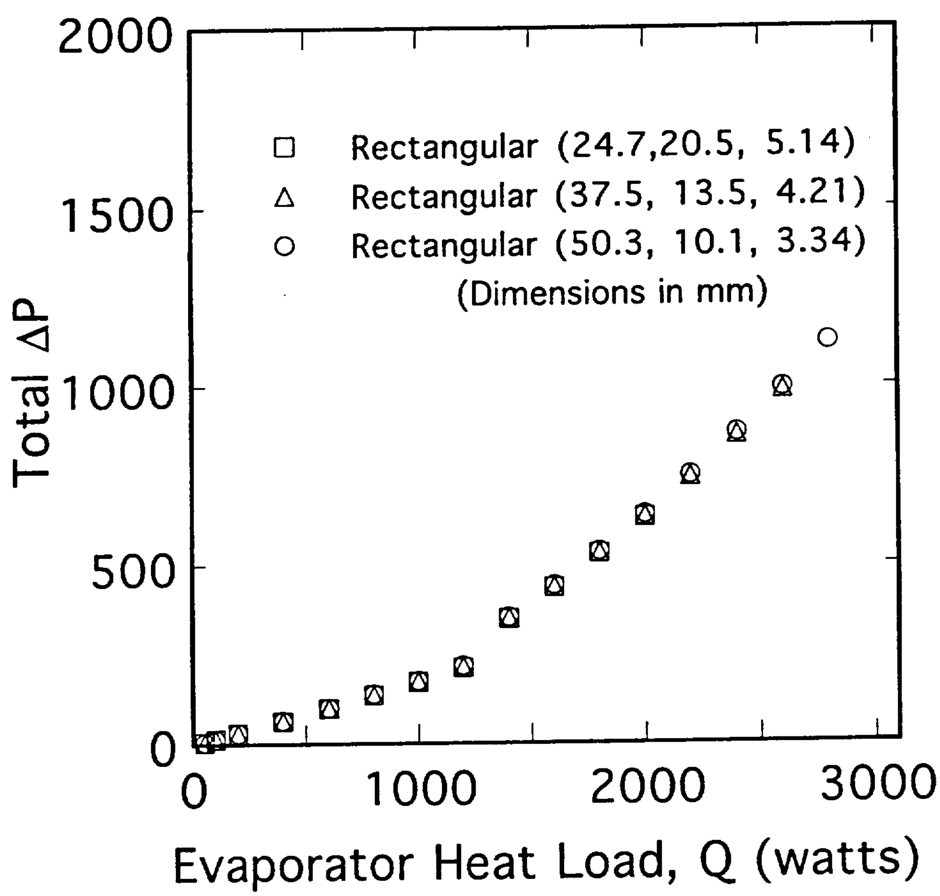

Figure 5. Effect of wick thickness on total pressure drop.

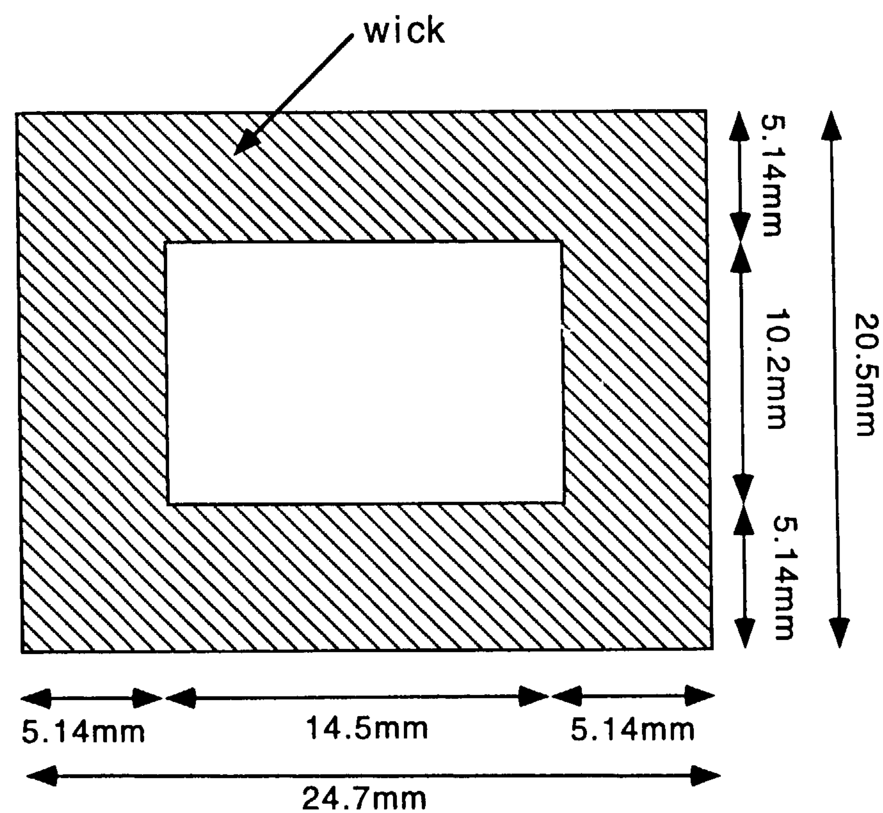

Figure 6. Dimensions for the 24.7 mm wide rectangular geometry.

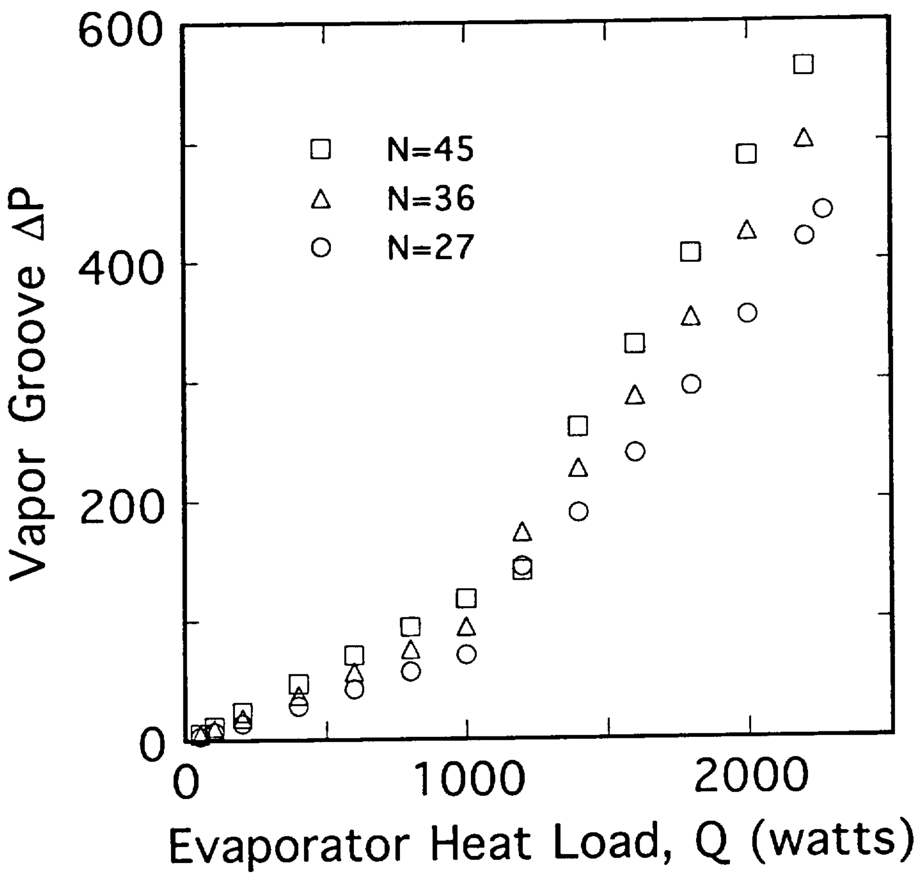

Figure 7. Effect of number of passages on vapor groove pressure drop.

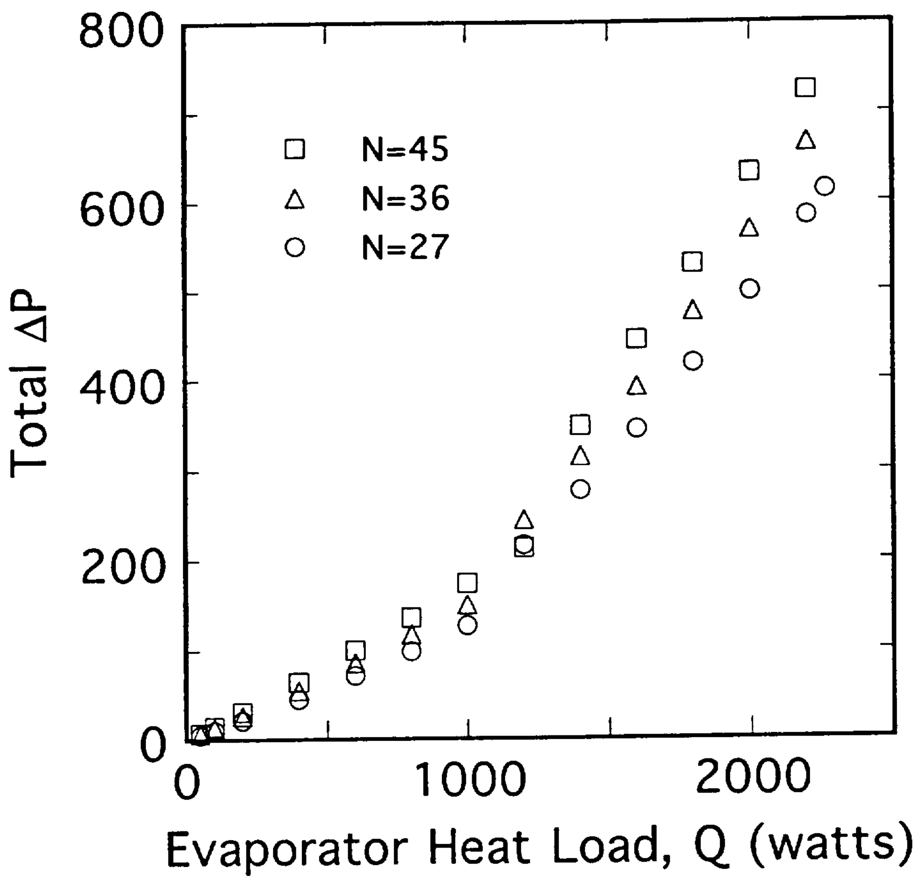

Figure 8. Effect of number of passages on total pressure drop

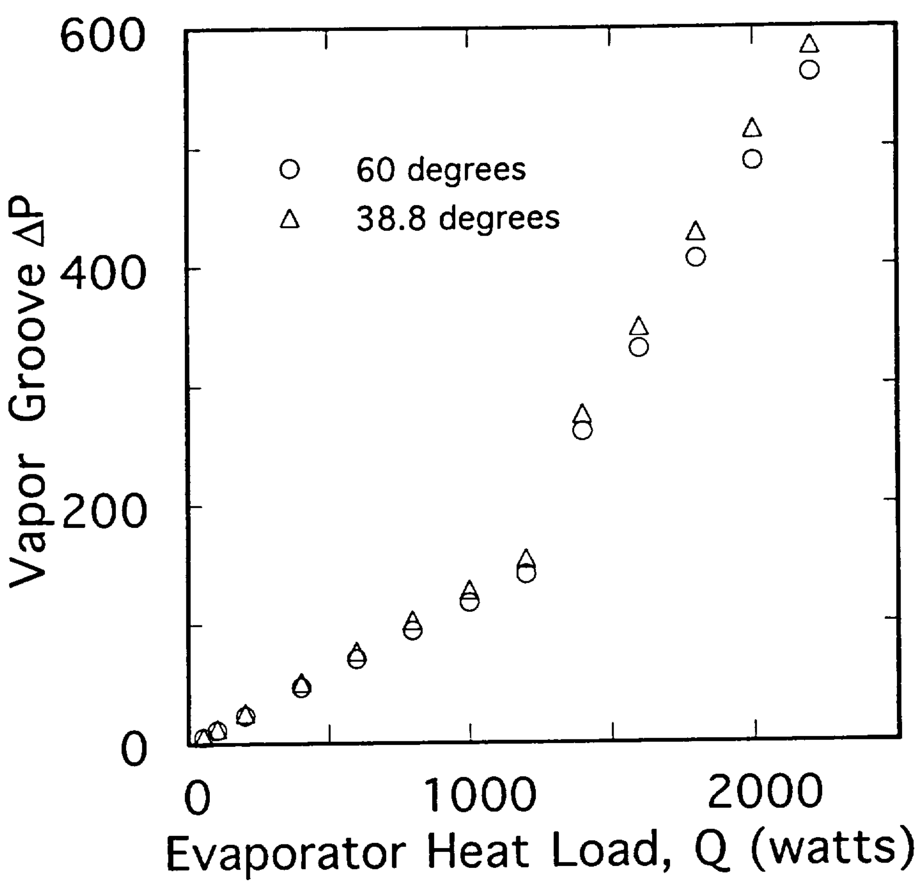

Figure 9. Effect of apex angle on vapor groove pressure drop

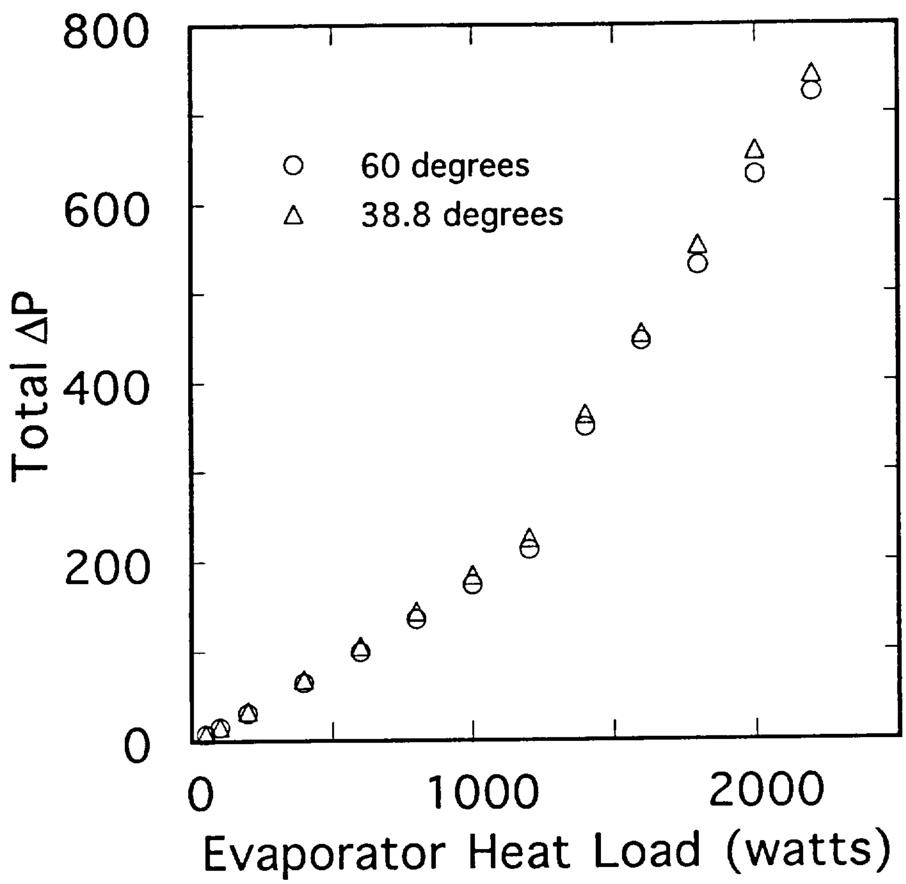

Figure 10. Effect of apex angle on total pressure drop.